NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 4023

DESIGN AND EXPERIMENTAL EVALUATION OF A LIGHT-WEIGHT TURBINE-WHEEL ASSEMBLY

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Lewis Flight Propulsion Laboratory
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SUMMARY

Light-weight design concepts were developed and tested in a full-scale turbine wheel with a tip diameter of 34.3 inches and a tip speed of 1190 feet per second. The rotor consisted of two thin A-286 alloy disks attached to each other by spacers and bolts with a separation of 0.625 inch. The disks were contoured for optimum stress distribution. The turbine blades were hollow castings of HS-21 (AMS 5385A) alloy with an integral serrated root.

The total weight of the wheel, exclusive of the shaft and bolts, was 121 pounds. The light weight, utilization of a disk material having a low strategic index, and simplicity of fabrication make the design particularly applicable to use in short-life expendable uncooled engines.

Failure occurred in a disk serration segment after 6 hours and 46 minutes of engine operation. Excessive blade vibration appeared to be the cause of failure.

INTRODUCTION

Research is being conducted at the NACA Lewis laboratory to investigate the possibility of reducing the weight and simplifying the production methods for turbine wheels. The decrease in weight is limited not only to the specific weight reduction effected in the wheel, but is reflected also in the reduction of the weight of the engine frame and aircraft frame made possible by the lighter wheel. Manufacturers' estimates of the ratio between the reduction in wheel weight and the resultant reduction in engine and aircraft frame weight range from 1:5 to 1:10. Three basic advantages are gained by weight reduction: (1) an increase in either range or load capacity of aircraft, (2) a decrease in the rotational inertia, which results in more rapid engine acceleration and less gyroscopic effect, and (3) conservation of strategic materials.

Simplication of production methods is emphasized particularly in reference to the heavy forge capacity that would be required to produce conventional turbine disks in large numbers in the event of military emergency. An attempt was made, therefore, to utilize rolled plate for the disks and thereby reduce the requirement for specialized fabrication methods.

Previous research in the program of wheel-weight reduction consisted of an investigation of several design variations of hollow uncooled turbine blades (ref. 1). One of the more promising designs stud- 4123
ied was a cast hollow blade with an integral serrated root designed in accordance with information contained in reference 2. This blade design was incorporated in a full-scale wheel; and this report will describe the design concepts used in arriving at the final wheel configuration. The complete wheel was fabricated and operated in a turbojet engine.

For clarification, the term "wheel" will be used in this report to denote the entire assembly, whereas the term "disk" will be used to denote the part of the wheel that is separate from the blades and shaft. The term "blade weight" refers to the actual weight of the turbine blade plus that serrated part of the disk rim associated with each blade base.

DESIGN CONCEPTS

Light-Weight Turbine Blades

As mentioned in the INTRODUCTION, the initial part of this research program was concerned with the development of design concepts for decreasing the weight of the blades. The results of this part of the program are reported in reference 1. Both sheet-metal and hollow-casting techniques were employed with variations in materials, wall thickness, attachment of airfoil to base, and methods of fabrication. It was shown that, through the use of the proper materials and construction techniques, both the sheet-metal and hollow-cast blades could be made to provide the required degree of reliability. The weight of these blades was roughly half the weight of equivalent solid blades. The serrated form of mechanical attachment of the blade bases to the disks was utilized in all phases of this part of the program to circumvent the problems inherent in attempting to use brazing or welding as a root fastening.

Effect of Rim Loading on Wheel Weight

The magnitude of the centrifugal force exerted by the turbine blades on the rim of the disk determines to a large extent the total weight of the wheel assembly. To illustrate this point, a wheel configuration and operating conditions were assumed for a constant disk-center stress, and the total wheel weight was computed as a function of the blade weight. A turbine tip diameter of 34.3 inches was used with a tip speed of 1190 feet per second. A radial temperature gradient was assumed with a center temperature of 520° F and a rim temperature of 1060° F; the temperature was assumed to vary as the cube of the radius between the center and the rim. For simplicity of computation, a parallel-sided disk was used; but, for more efficient use of the disk material in a practical application, the sides of the disk would be contoured.

The results of the computation are shown in figure 1 for the three arbitrarily chosen disk-center stresses of 80,000, 90,000, and 100,000 pounds per square inch. For the wheel design chosen, the solid-blade weight would be approximately 0.6 pound, leading to a total wheel weight of 340 pounds for a disk-center stress of 80,000 pounds per square inch. If, through the use of a hollow-type blade construction, the blade weight could be reduced by 50 percent, the total wheel weight could be reduced 170 pounds (for the same disk-center stress). The total wheel-weight reduction of 170 pounds therefore would have resulted from a blade-weight reduction of only 28.8 pounds or a 56.5-pound reduction in wheel weight for every 0.1-pound reduction in individual blade weight. It is evident that the blade weight should be reduced as much as possible, consistent with strength requirements.

Selection of Disk Material and Its Effect on Wheel Weight

If it is assumed that the blade configuration and, hence, the rim loading have been established, it is necessary to select a disk material and a suitable disk contour. Two factors are involved in the dependence of the wheel weight on the disk material: (1) the physical strength properties of the material, and (2) the degree of utilization of these properties. Since the center of the disk operates at a temperature of approximately 400° to 500° F, the yield strength and ductility are prime considerations in this region. However, the rim of the disk, particularly the serrations, operates at temperatures that make it necessary to consider creep and time-to-rupture.

Consideration was given to composite disk construction, utilizing different materials in the rim area and in the center. This type of construction would permit more latitude in selection of materials, but it would introduce problems of welding not within the scope of the present report. In addition, the use of a single material is preferable from the standpoint of simplification in production methods, an important objective of the research program. It is probable that future research and development will eliminate the problems of composite construction in this application.

It is therefore necessary to select a single material that will have both high short-time yield properties at low temperatures and high stress-rupture properties in the 1000° to 1200° F temperature range. These two requirements severely limit the selection of material. In addition, since it is preferable to start with rolled plate, as stated previously, the selection of material is further limited. 4123

The gains that can be obtained by increasing the permissible center stress in the disks through the use of materials of higher strength are indicated in figure 1. For example, for a blade weight of 0.4 pound, the total wheel weight would be reduced from 227 pounds with a center stress of 80,000 pounds per square inch to 132 pounds with a center stress of 90,000 pounds per square inch (a weight reduction of 41.8 percent). These figures were obtained on the basis of a parallel-sided disk, as noted previously.

The other factor involved in the dependence of the wheel weight on the disk material is the degree of utilization of the physical properties of the disk. For maximum utilization it would be preferable to operate the disk with the stress level at any given radius equal to the controlling property at that radius. It is obvious, however, that some factor of safety must be employed. The magnitude of the safety factor will depend on the reliability of the physical properties of the raw material, the effect of the machining and heat treatment in producing variation in properties, the influence of stress concentrations, and the lack of knowledge as to the actual stress history to which the wheel may be subjected. Each specific design should therefore be evaluated individually to arrive at a logical value for the safety factor.

In addition to the factors stated, there are other considerations to be evaluated in selecting a disk material, such as: machinability, response to heat treatment, fatigue strength, strategic index, stress corrosion, and wear.

Rim Width and Disk Contour Determination

The determination of the rim width is dependent primarily on the allcwable shear and tensile stresses in the blade-fastening area. A study of the strength of the serration-type mechanical fastening as influenced by such factors as time, temperature, material, and configuration is reported in reference 2. By using this information, in addition to the factor of safety required by such unknowns as vibration, bending, and thermal stress, a nominal rim-width value may be obtained.

The required rim width having been established, it is next necessary to determine the proper disk contour to complete the basic wheel design. This can be accomplished through the use of an allowable-stress curve, determined as described previously, in conjunction with the direct method of design of rotating disks outlined in reference 3. A disk contour that utilizes the material in the most efficient manner and produces the lightest disk under the given assumptions will result directly from the computation.

When disk-contour computations that use rim loadings associated with the lighter sheet-metal or hollow-cast blades were made, it was found in most instances that the disks became quite thin (of the order of 0.2 to 0.4 in. thick). As a result, several additional problems arose in the final determination of the design configuration. One of these problems is involved in the method of transferring the centrifugal and bending loads of a relatively wide blade to a relatively thin rim. The other problem arises as a result of the deficiency in stiffness of a thin disk when subjected to forces perpendicular to the disk face. These forces may be induced by gyroscopic effects, periodic vibrational disturbances, and uneven temperature distributions. One method of circumventing these problems for a single-stage turbine is the use of two thin disks spaced to provide optimum blade-load distribution and bolted together to increase the lateral stiffness. In the case of multistage turbines, the individual disks of the various stages could be tied together and thereby the lateral stiffness improved.

DESIGN APPLICATION

The concepts described in the previous section were used to design a light-weight, uncooled turbine wheel that was fabricated and operated in a turbojet engine. The spaced double-disk principle was used to provide the required lateral stiffness, and the standard type of root serration was used for the blade fastening.

Turbine Blades

On the basis of research reported in reference 1, the cast hollow-blade configuration shown in figure 2 was chosen. The 96 blades were made by the lost-wax casting process with a wall thickness of 0.055 inch at the base, tapering linearly to 0.030 inch at the tip. The material of the blade was as-cast HS-21 (AMS 5385A). The suction and pressure sides of the blade were joined by an integral spanwise rib in the tip area to minimize vibration problems. The airfoil required no finishing operations; the root serrations were crush-ground and shot-peened. The centrifugal tensile stress at the base of the airfoil was 21,000 pounds per square inch at rated speed. The finished blade (fig. 3) weighed 0.34 pound.

Rim Width

On the basis of the information contained in reference 2, the rim width for each of the two disks was computed to be 0.310 inch for a serration shear stress of 27,900 pounds per square inch and a tensile stress at the bottom of the disk serration segment of 35,400 pounds per square inch. The allowable stress values were based on the use of A-286 as the disk material, which was chosen in accordance with the requirements listed in the section of this report entitled "Selection of Disk Material and its Effect on Wheel Weight." 4123

Disk Contour

Establishment of a curve of allowable-stress against radius was based on a predicted radial temperature profile. This profile was obtained by using information obtained at the Lewis laboratory by C. R. Morse and J. R. Johnston during investigation of engine-operating temperatures, in which standard turbine-wheel temperatures were measured under various conditions of operation and with various means of disk cooling. The predicted radial temperature profile is shown in figure 4 with the center temperature at 505° F and the bottom of the serration at 995° F. Using manufacturers' published data on the physical properties of A-286 and a safety factor of 1.25, the allowable stress was plotted against the disk radius as shown in figure 5. The physical properties chosen as controlling the allowable stress were the 0.02-percent short-time yield and the 0.1-percent creep in 1000 hours. It will be noted that, for the material and the particular set of conditions imposed, the controlling factor from the center of the disk to the serration is the 0.02-percent short-time yield.

The rim width was assumed constant at 0.310 inch from the 12.20-inch radius to the outside diameter of the disk. On the basis of the established allowable-stress curve, the disk contour was computed by the method of reference 3, and is shown in figure 6(a). The thickness at the center of the disk is 0.397 inch, and tapers to 0.166 inch at the 11.59-inch radius. The computed stresses for this profile at a rated engine speed of 7950 rpm are shown in figure 6(b). It will be noted that the equivalent-stress curve is identical with the allowable-stress curve in figure 5 except in the extreme rim region. Here the computed stresses are lower than the allowable stresses because of the necessary transition from the thin neck of the disk to the required rim width.

Final Disk Designs

The idealized contour of figure 6 was modified to provide for attachment of the disks to each other and to the shaft and to simplify the machining process. The central portion of the disks, out to the 4-inch

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radius, was made parallel-sided with a thickness of 0.380 inch. From the 4-inch radius, the thickness was reduced linearly to 0.190 inch at the 11.79-inch radius and then was increased in a generous radius to meet the 0.310-inch rim thickness. In the machining process, the entire contouring was put on one face of each disk, and the plane surface was retained on the other face. Figure 7 shows the final design configuration with the disks spaced 0.625 inch apart for optimum rim-loading distribution. This spacing was determined by considering the blade in several axial sections and then placing the points of support in lines radial with the center of gravity of the respective sections. A circle of five bolt holes was used at the 3-inch radius to provide for attachment to the flanged end of a turbine shaft. A circle of 15 through-bolts was installed at the 10.5-inch radius with appropriate spacers to provide for additional lateral stiffness. Figure 8 shows the comparison between the computed theoretical disk profile and the actual test disk profile.

As a final step in the design, the disk stresses of the modified computed profile were determined and are plotted in figure 9. In the region of the bolt holes, the disk thicknesses used in the stress computation were reduced in proportion to the amount of material removed, thus providing an indication of the increase of the average stresses resulting from the holes. The equivalent-stress curve is still essentially the same as that shown in figure 6(b).

Machining of the disks from the A-286 rolled plate was accomplished in the solution-treated condition ($1800^{\circ}\pm25^{\circ}$ F for 1 hr and air-cooled). After machining, the disks were aged at $1325^{\circ}\pm25^{\circ}$ F for 16 hours and air-cooled. A photograph of the completed wheel assembly is shown in figure 10. The total weight of the wheel was 121 pounds. The central disk shown in the photograph was provided for balancing purposes.

ENGINE OPERATION AND DISCUSSION OF RESULTS

The completed wheel was installed in a turbojet engine and operated in a sea-level test stand. The wheel was operated at intermediate speeds of 6000 and 7000 rpm for 1/2 hour each. Subsequently, the wheel was operated at maximum rated operating conditions (7950 rpm and 1260° F exhaust temperature) for runs of $\frac{1}{2}$- and $1\frac{1}{2}$-hour duration, respectively. After each of the runs, the wheel was visually inspected and measured. At this point in the operation schedule there were no detectable signs of trouble. Operation was resumed at rated speed and temperature until failure of a disk serration segment occurred after a total running time of 6 hours and 46 minutes. The failure is shown in figure 11. Examination of figure 11(b) indicates that the fracture was progressive.

Since the computed stresses were well within the allowable-stress limits in the serration area, an investigation was made to determine the cause of the premature failure. The properties of the disk material were checked by cutting specimens from the disks themselves. Chemical analysis, hardness, microstructure, smooth and notch stress to rupture, and ductility were investigated. All these tests indicated that the properties of the material itself were actually somewhat superior to the published data. On the basis of this information it was concluded that the failure was due to loading of the serrations in a manner that cannot be handled adequately in the design analysis. The possibility of excessive vibration of the blades was therefore considered, particularly because of the progressive nature of the fracture surface. This type of disk failure is not unique; similar serration failures have occurred in standard turbine wheels.

Resonant frequencies and nodal patterns for the blades were determined and are shown in figure 12. The blades were inserted in a turbine wheel modified to provide tight base clamping. A speaker-type unit was used to excite vibration. Nodal patterns were determined with a crystal cartridge pickup. Correlation of these data with some of the excitation orders determined in reference 4 for the test engine are shown in the speed-frequency diagram of figure 13. There were a number of possibilities for blade resonance in the speed range covered by the schedule of operation of the light-weight wheel. To determine whether it would be possible to overstress the wheel serration segments by the suggested blade vibration, a mockup of the wheel and blades was made as shown in figure 14. Portions of the disks were welded to a heavy base plate and alined to simulate the actual wheel configuration. Two of the hollow cast blades were inserted in the serrations and centrifugal force was simulated by pushing upward on the blades by means of bolts through the base plate. A strain gage was mounted at the base of each airfoil in the midchord of the suction surface.

The blades of the mockup were vibrated in the fundamental bending mode by excitation with a jet of air interrupted by a slotted disk rotating at the proper speed to provide resonance. Within approximately 5 minutes, the stress at the location of the gages dropped from ±12,000 to ±8000 psi with a constant air pressure, and the resonant frequency dropped 60 cycles per second. The mockup was dismantled and subjected to a penetrant-oil inspection. Two cracks were observed in the wheel serration segments, as shown in figure 15. These photographs are combination natural-light and black-light exposures showing the fluorescent-oil indications of the cracks. The type of failure obtained in the mockup was similar to that obtained in the engine.

The failure in the turbine wheels was confined strictly to the serration region. Measurements of the disks themselves showed no deformation (within usual limits of measurement accuracy) and examination of the blades showed them to be free from cracks (penetrant-oil inspection). On the basis of this analysis it is believed that either the rim could be redesigned to resist undue blade vibration or changes in turbine geometry could be made to reduce unfavorable excitations.

SUMMARY OF RESULTS

Design concepts were discussed for obtaining turbine wheel configurations that have the advantages of light weight, simplicity of fabrication, and lowered strategic alloy content. It was shown that the total wheel weight is critically dependent on the blade weight and on the degree of utilization of the strength available in the disk material through proper contouring.

On the basis of these concepts, a light-weight, single-stage, turbine wheel design was evolved and fabricated. The 96 blades were hollow castings of HS-21 (AMS 5385A) alloy weighing 0.34 pound each and were attached to the disks by a serration-type fastening. Two contoured disks were employed, spaced 0.625 inch apart to obtain better blade-load distribution and to increase the lateral stiffness. The disks were contoured on one side only and were machined from rolled plate to eliminate requirements for heavy forge capacity. The disks and blade assembly weighed 121 pounds, showing approximately a 46-percent reduction in weight compared with the standard wheel for the design chosen.

The wheel was bolted to a flanged shaft and installed in a turbojet engine. After 6 hours and 46 minutes of operation, a failure occurred in one of the disk serration segments. The failure occurred while the engine was being operated at rated speed and exhaust-gas temperature. An investigation of the failure indicated that the turbine blades may have been vibrating excessively, thus causing an overload of the disk serration segment. No deformation was apparent through measurement of the disks, and penetrant-oil examination of the blades showed them to be uncracked. It is believed, on the basis of the failure analysis, that either the rim could be redesigned to resist the excessive blade vibration or changes in turbine geometry could be made to reduce unfavorable excitations.

The light-weight wheel design proposed would be particularly applicable to an expendable missile engine.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, April 25, 1957

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2. Meyer, André J., Jr., Kaufman, Albert, and Caywood, W. C.: Investigation of Mechanical Fastenings for Solid Turbine Blades Made from Ductile Materials. NACA RM E54E21, 1954.

3. Manson, S. S.: Direct Method of Design and Stress Analysis of Rotating Disks with Temperature Gradient. NACA Rep. 952, 1950. (Supersedes NACA TN 1957.)

4. Morgan, W. C., and Morse, C. R.: Experimental Investigation of the Vibration Characteristics of Four Designs of Turbine Blades and the Effect Produced by Varying the Axial Spacing Between Nozzle Blades and Turbine Blades. NACA RM E51J25, 1952.

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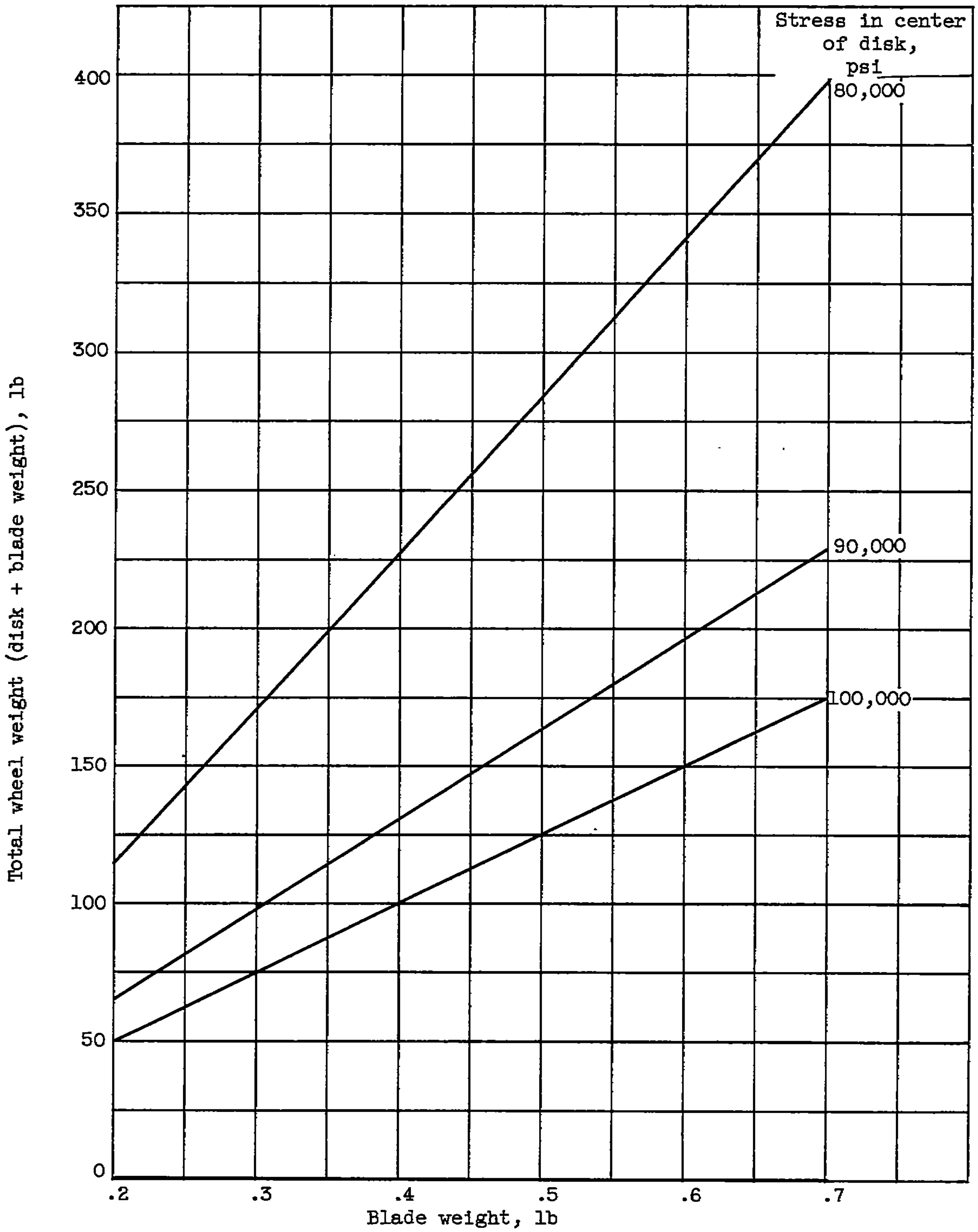

Figure 1. - Effect of blade weight on total wheel weight for several disk-center stresses. Parallel-sided disk assumed with a radial temperature profile of 520° to 1060° F.

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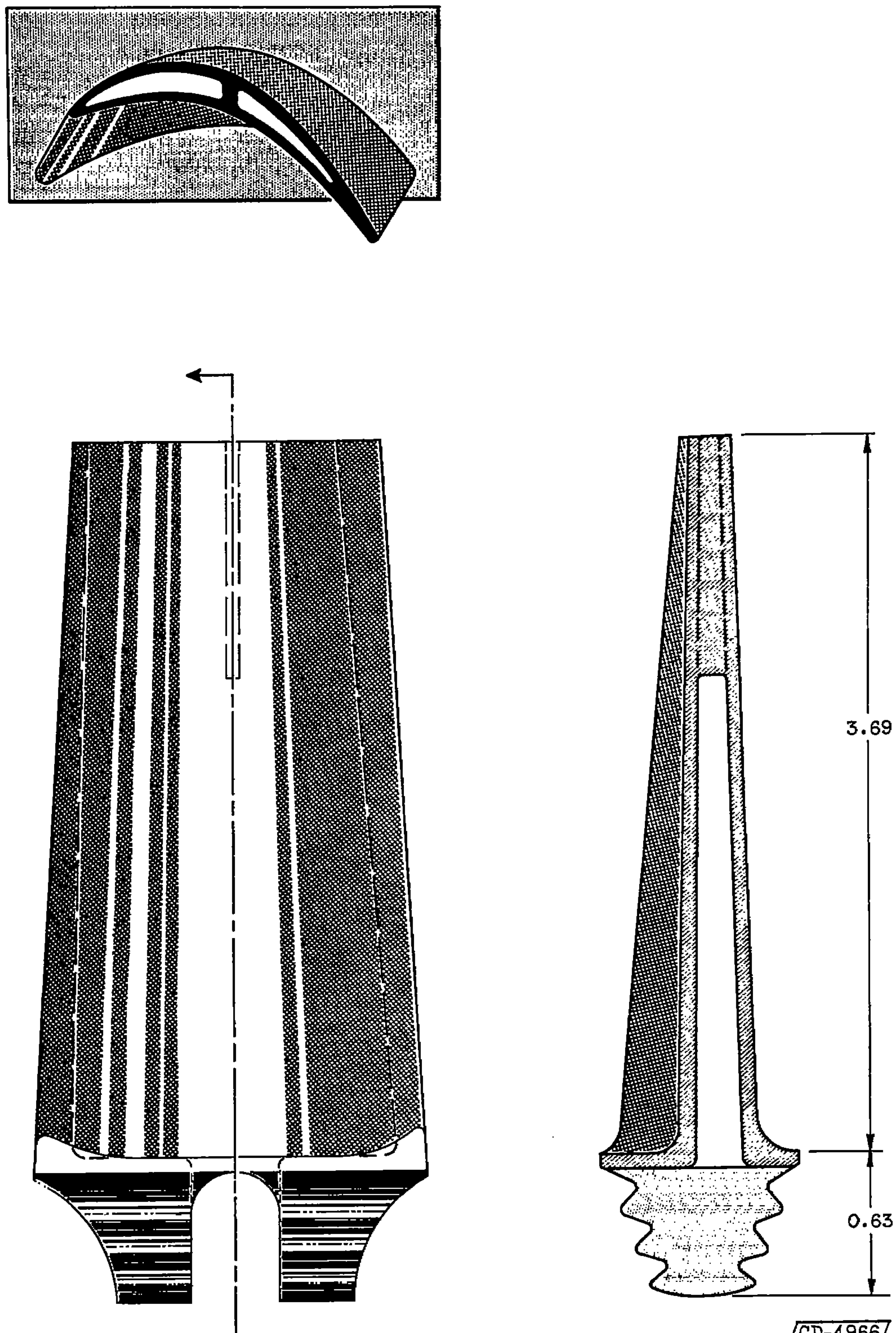

Figure 2. - Sketch showing construction of hollow turbine blade.

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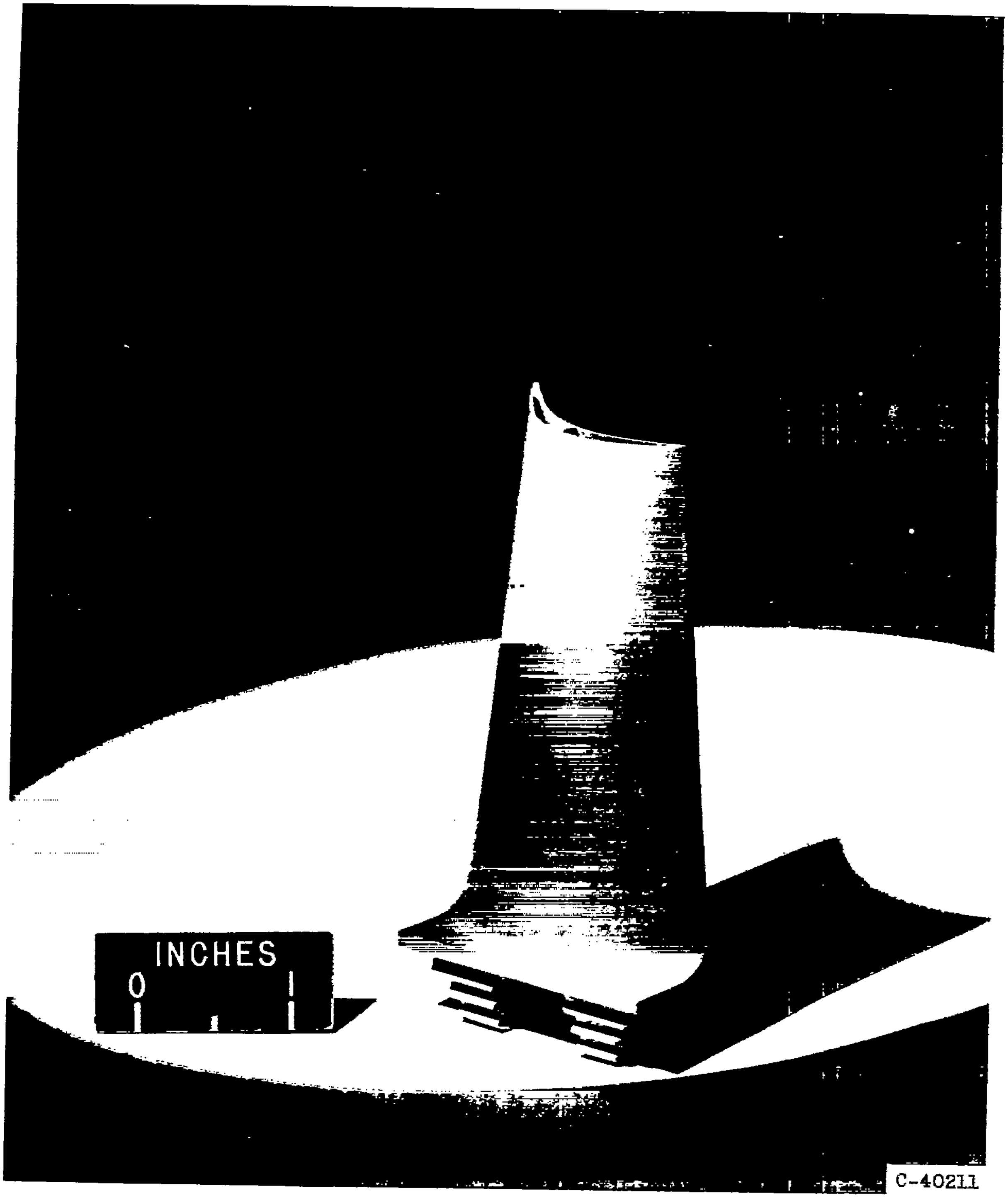


Figure 3. - Cast hollow turbine blade ready for installation.

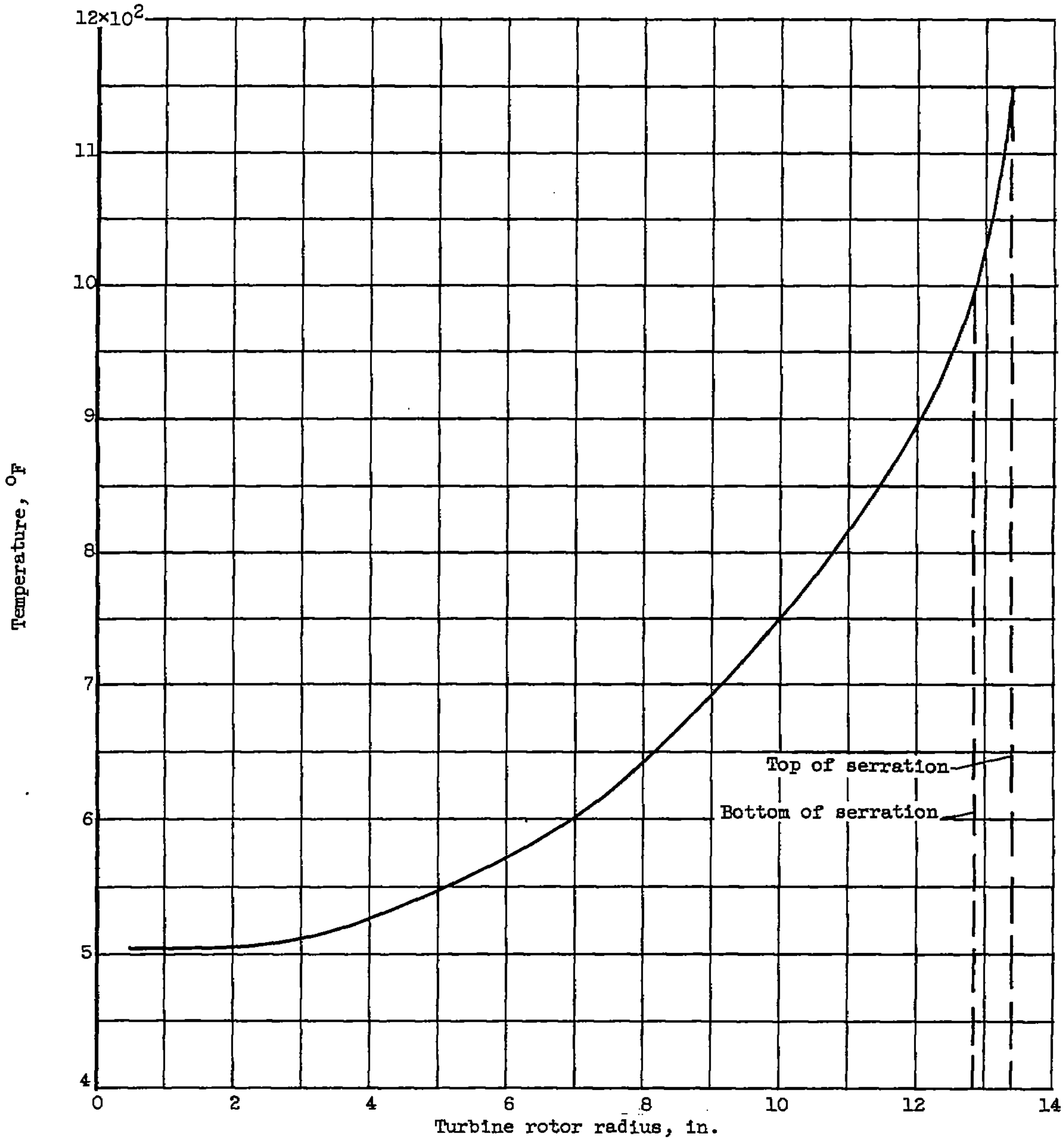


Figure 4. - Assumed operating temperature for light-weight turbine rotor.

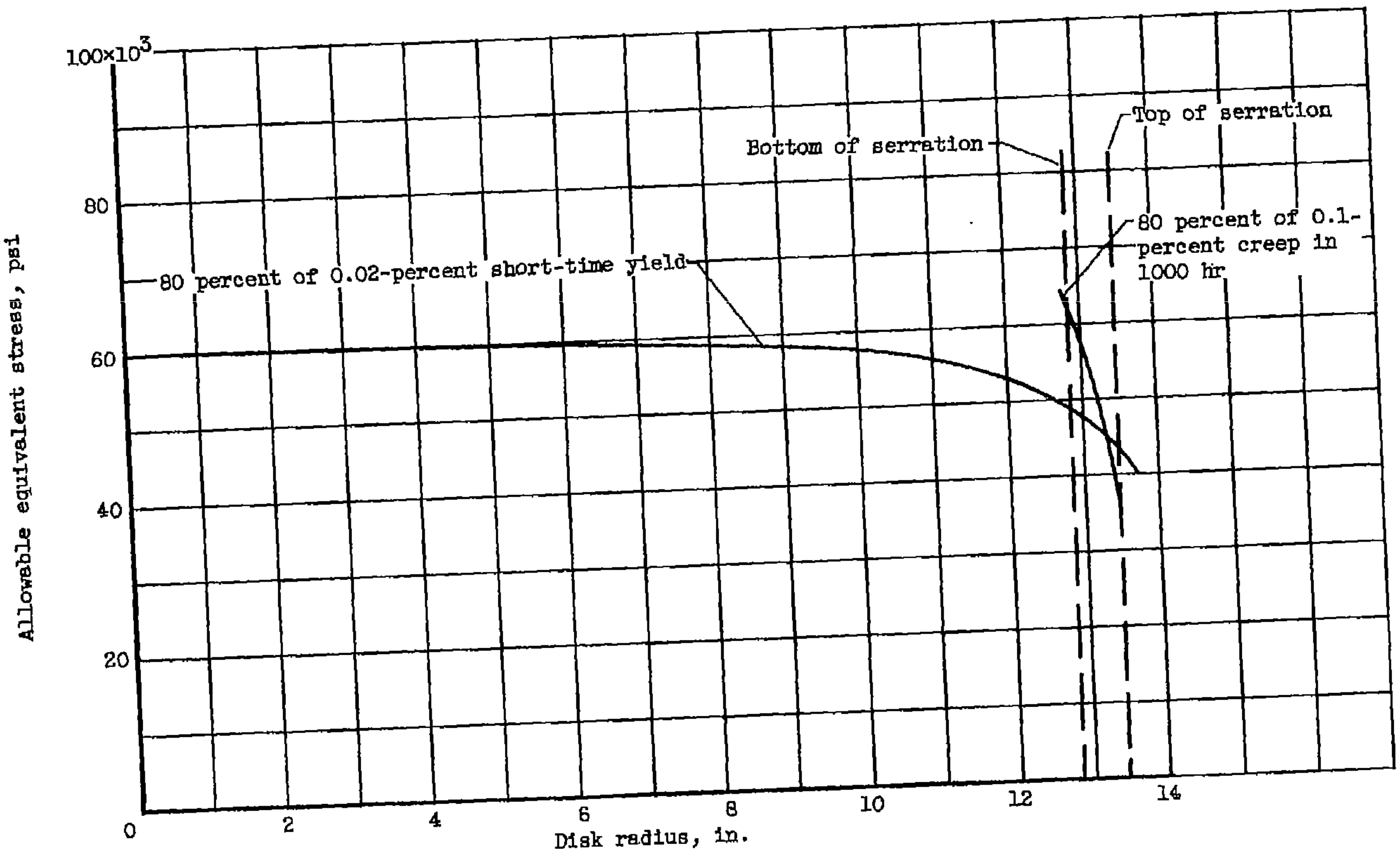
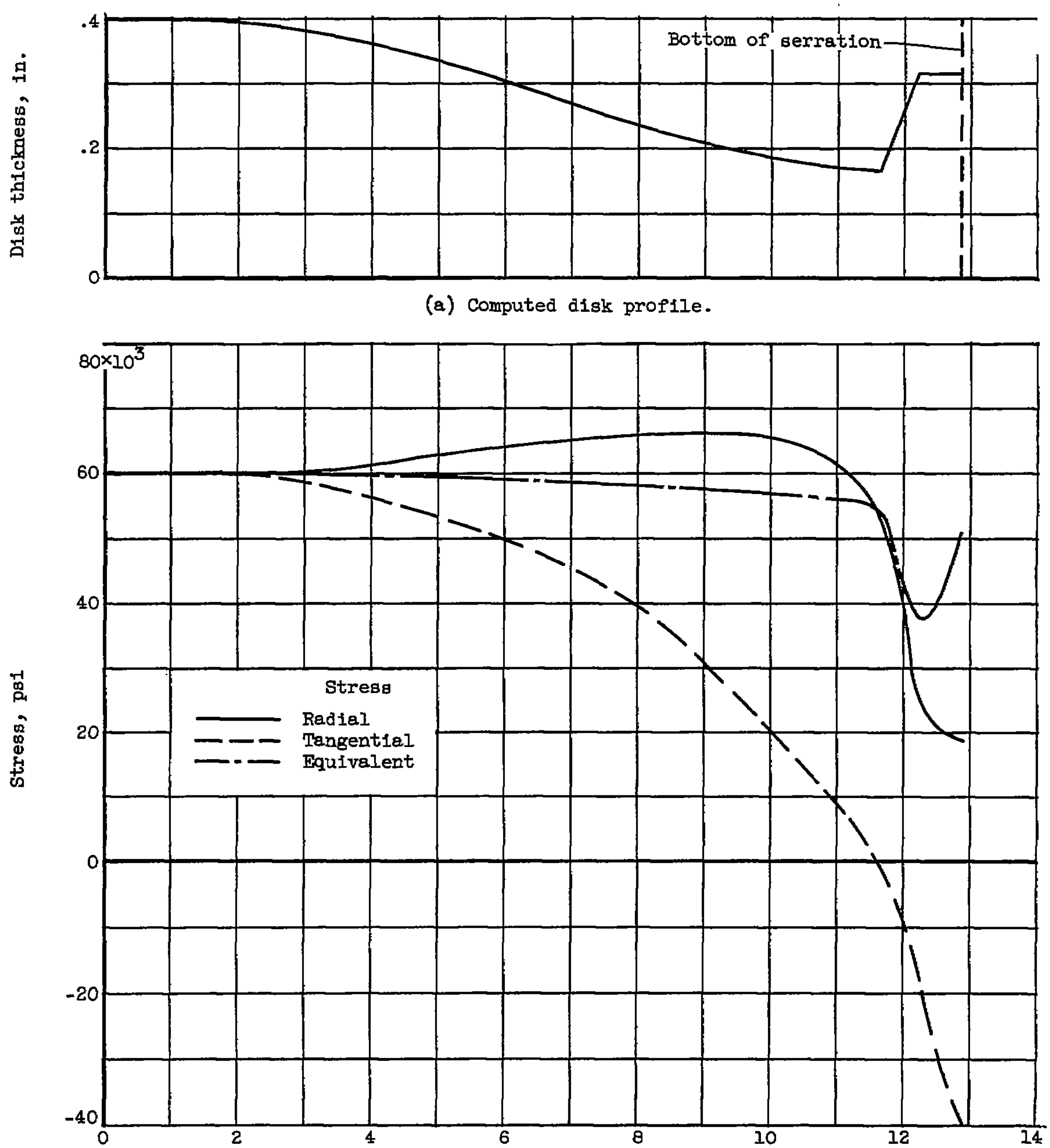


Figure 5. - Allowable disk stress based on predicted temperature profile and 80 percent of 0.02-percent short-time yield and 0.1-percent creep in 1000 hours.



(b) Stresses for computed disk profile at rated engine speed of 7950 rpm.

Figure 6. - Computed disk profile and stresses based on allowable stress curve of figure 5.

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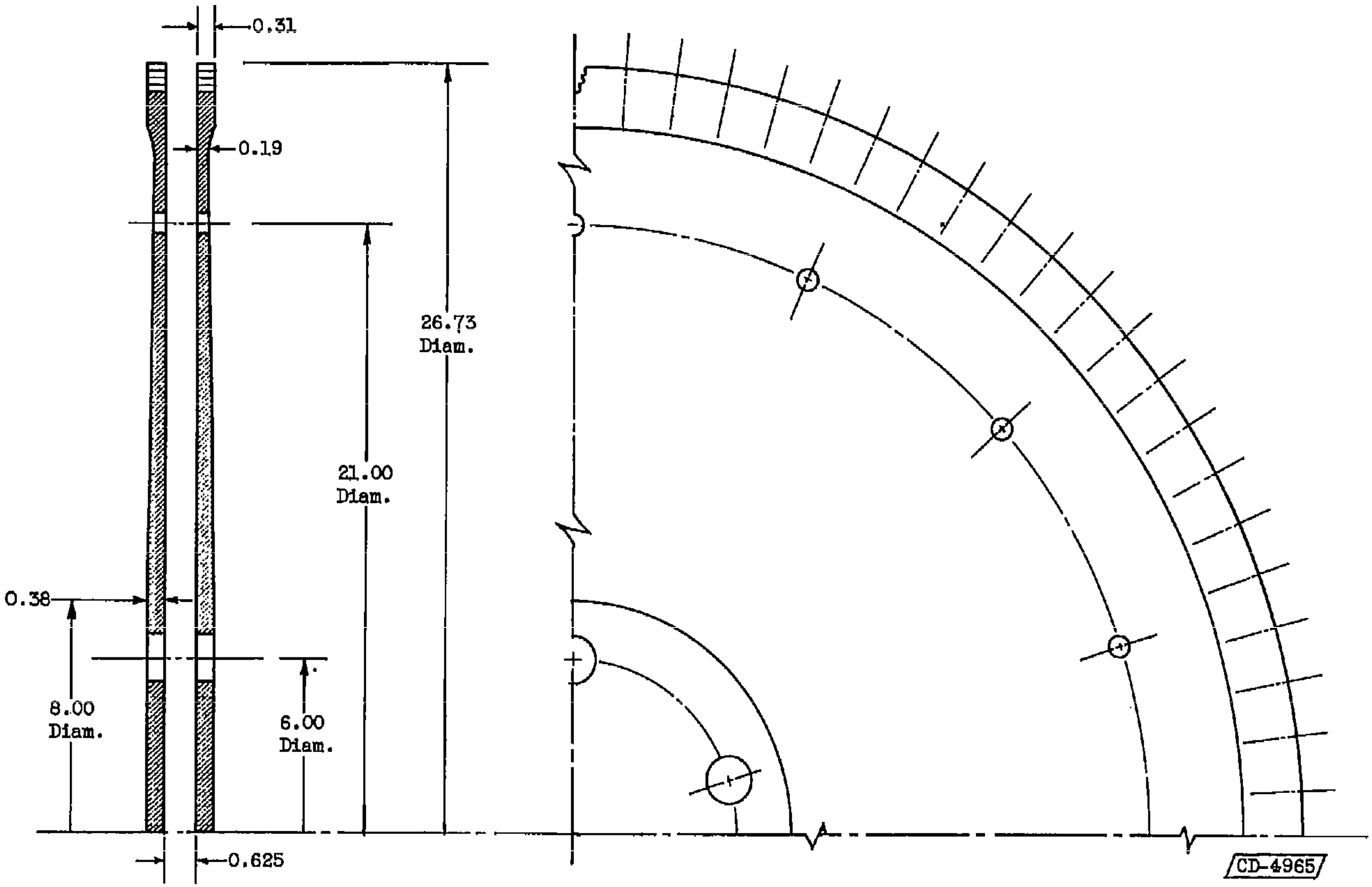

Figure 7. - Design configuration for light-weight turbine disks. (All dimensions in inches.)

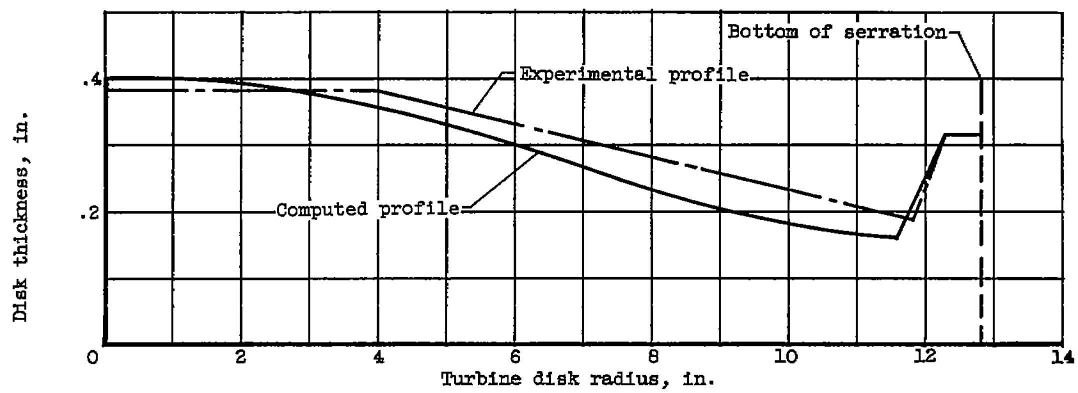

Figure 8. - Comparison between computed theoretical disk profile and actual experimental disk profile.

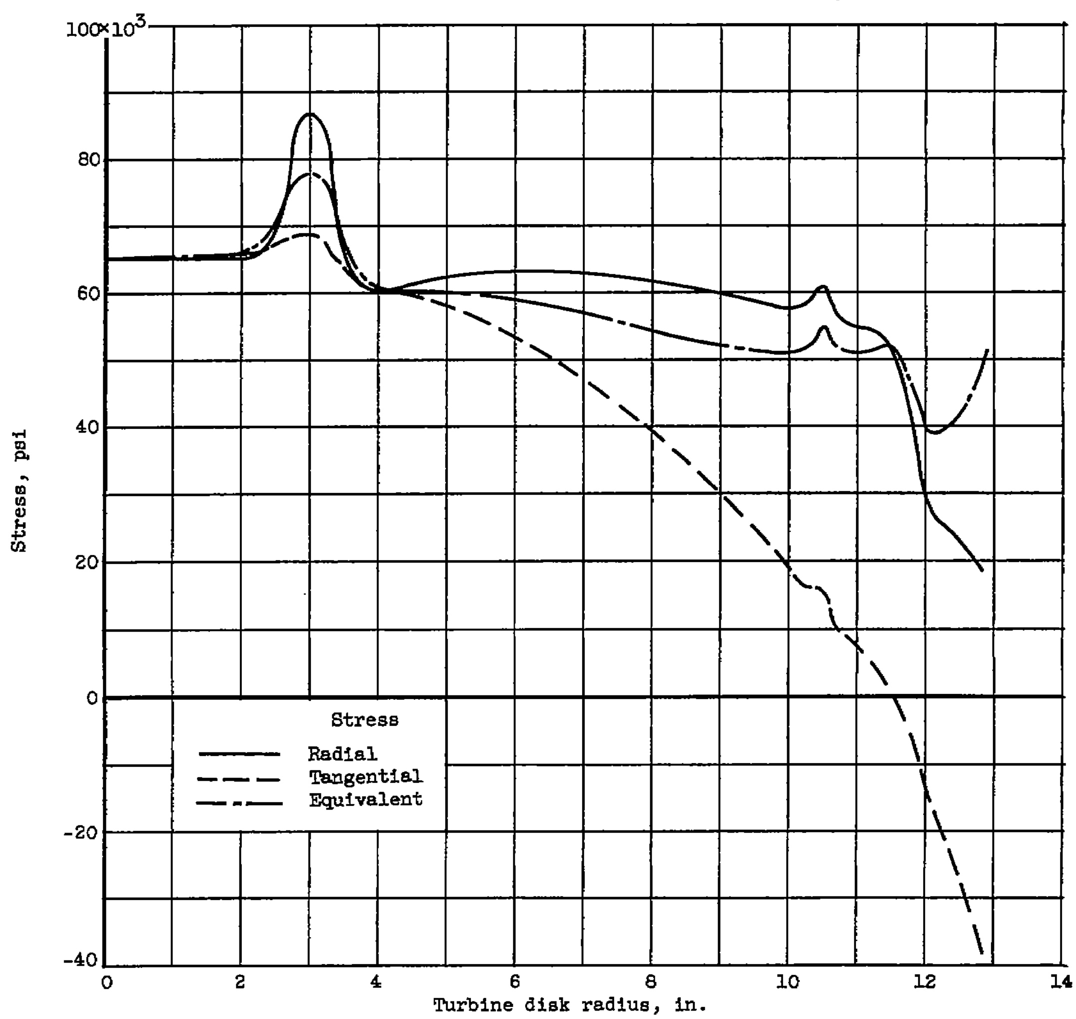

Figure 9. - Radial, tangential, and equivalent stresses calculated for light-weight turbine disks at 7950 rpm and thermal gradient shown in figure 4.

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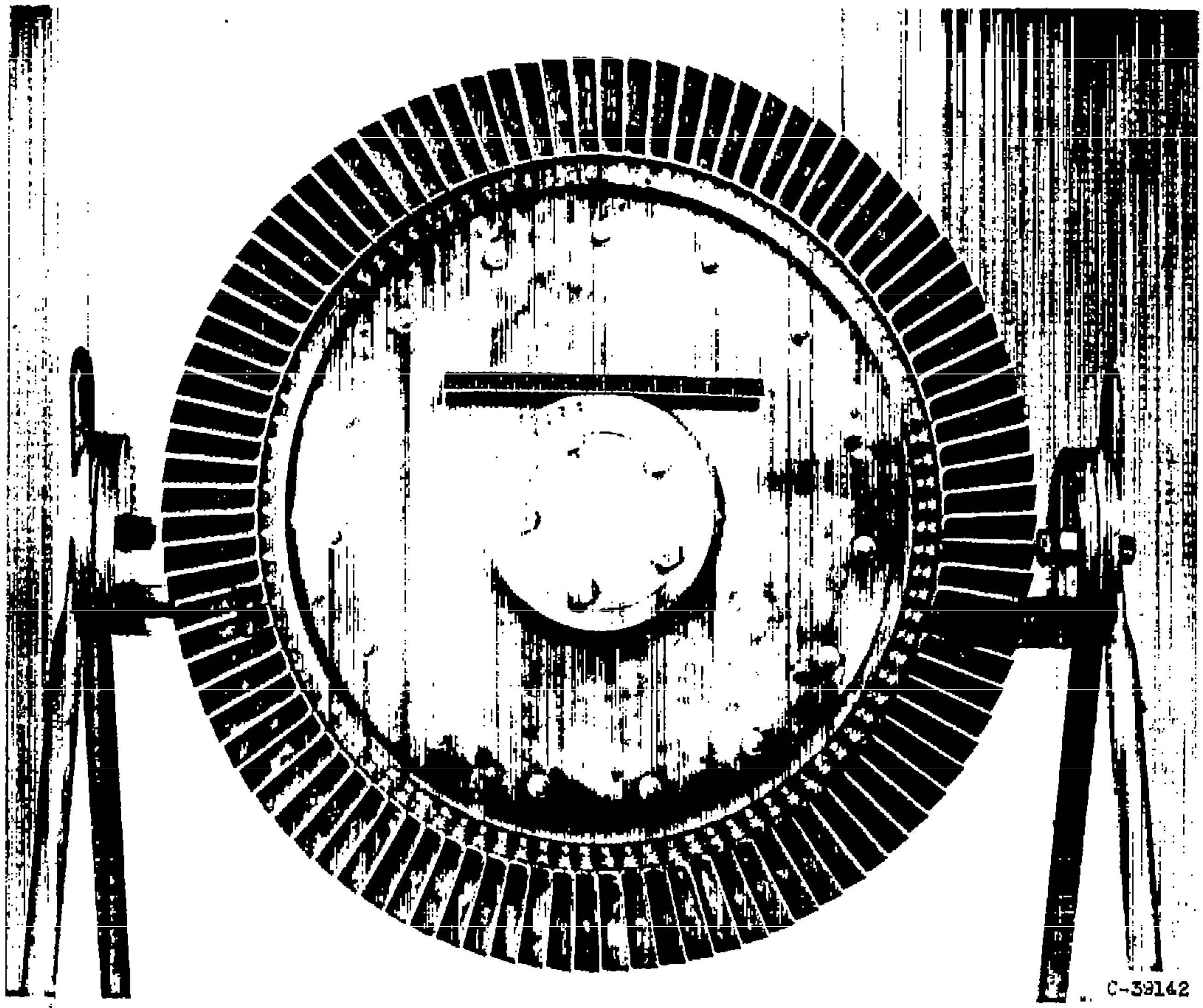

Figure 10. - Light-weight turbine wheel assembled and ready for installation.

(a) Over-all view.

Figure 11. - Failure of disk serration segment.

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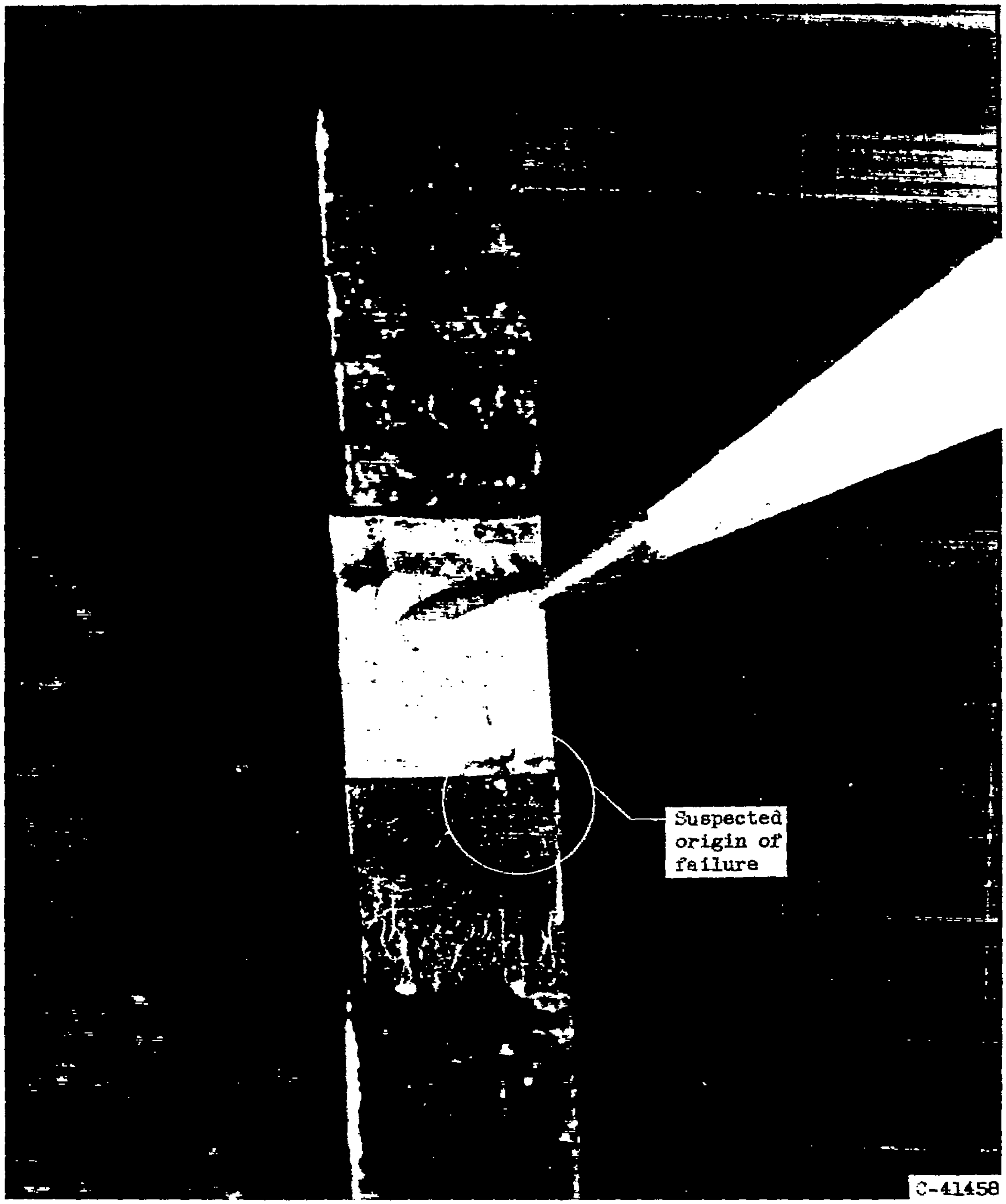

(b) Enlarged view.

Figure 11. - Concluded. Failure of disk serration segment.

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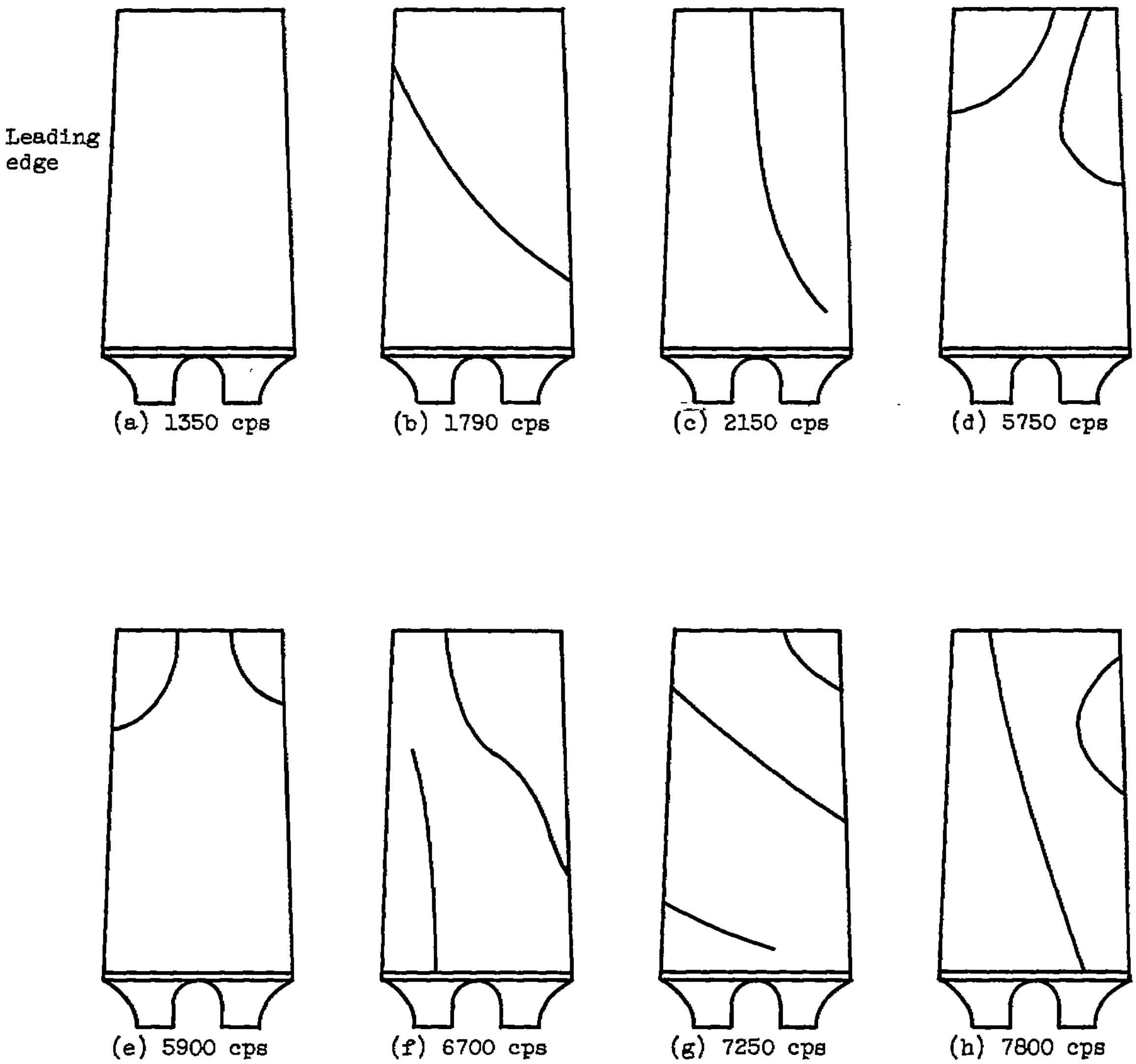

Figure 12. - Nodal patterns of cast hollow-blade vibration modes.
Patterns approximately the same on suction and pressure sides.

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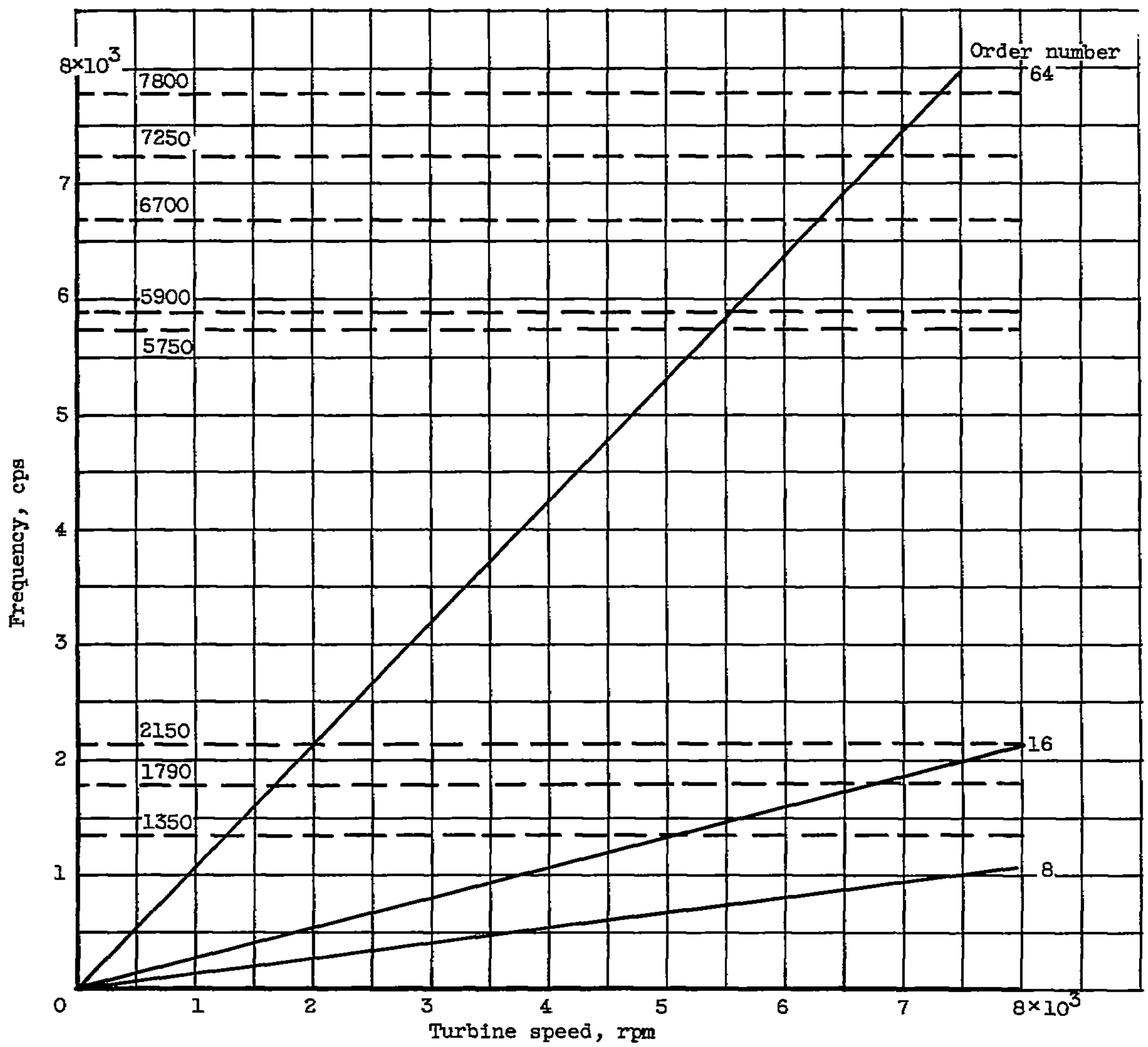

Figure 13. - Speed-frequency diagram showing blade resonant frequencies and probable important excitation orders.

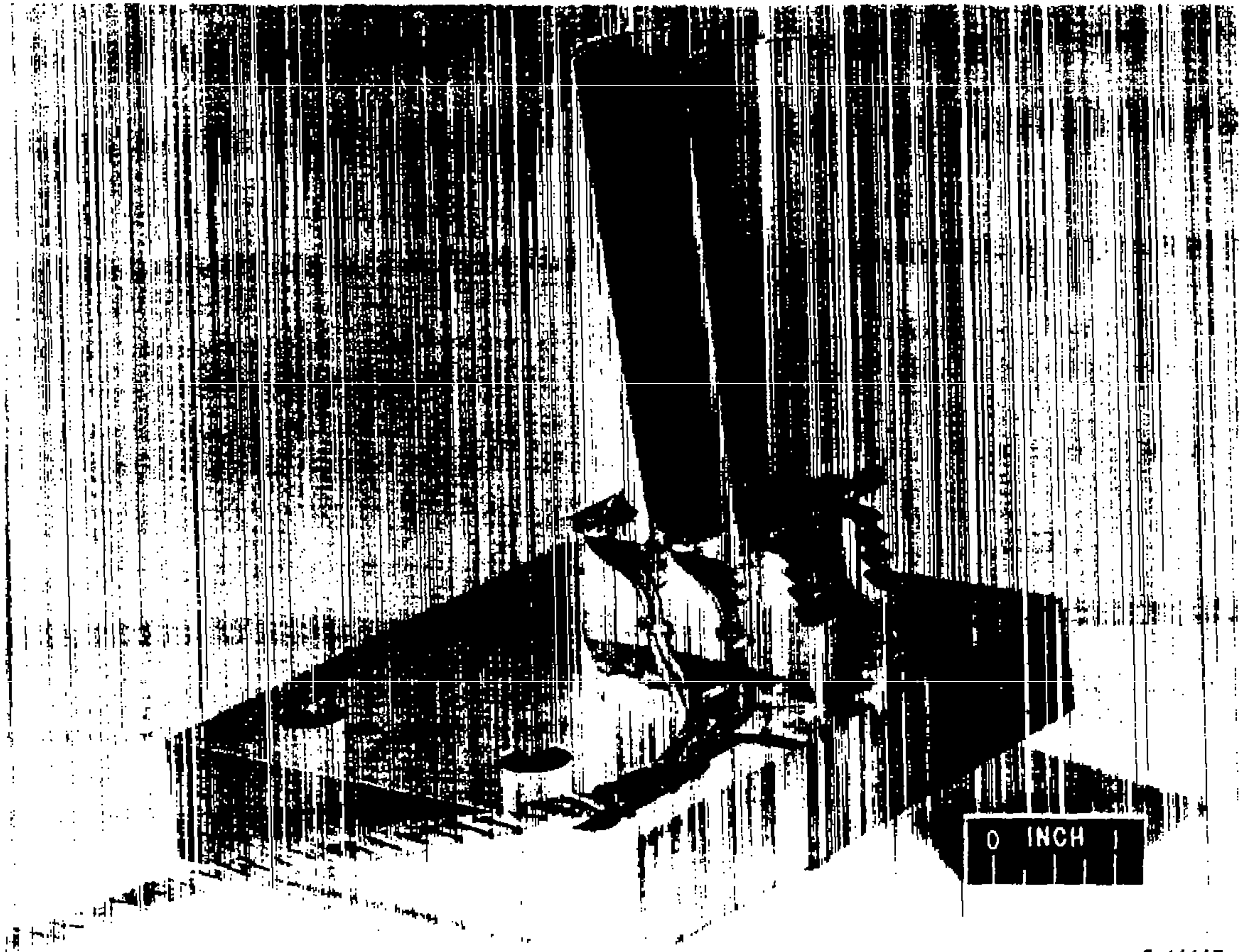

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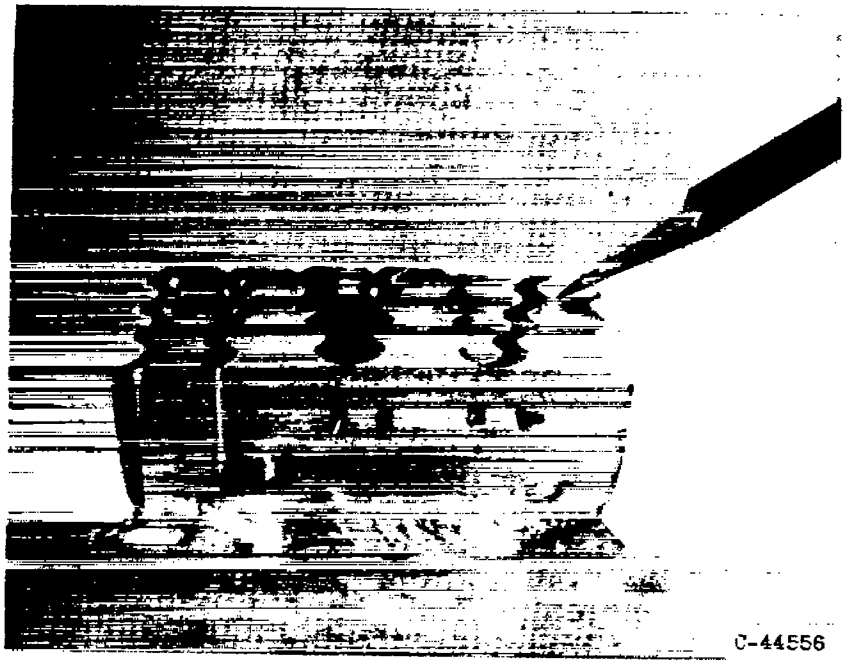
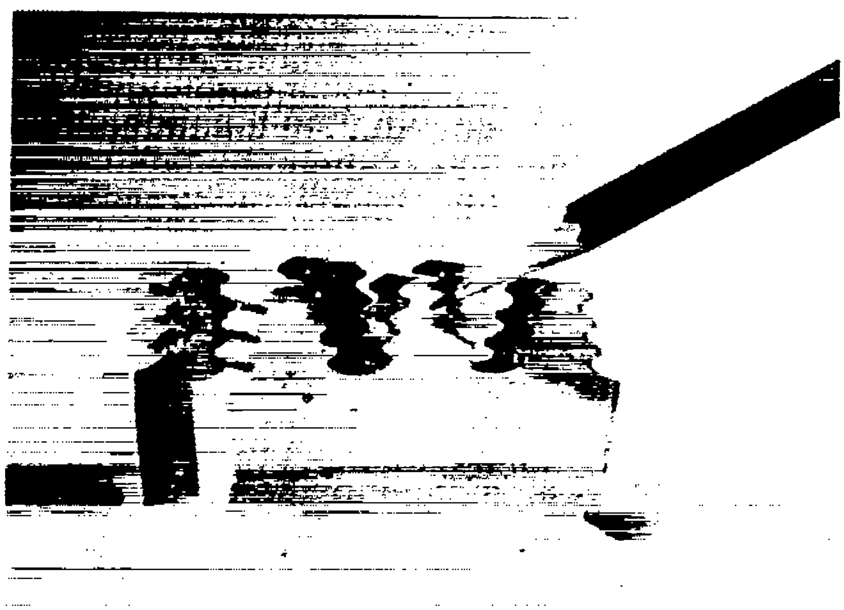
Figure 14. - Test setup of wheel segments and blades for vibration by interrupted air jet.



(a)



(b)

Figure 15. - Failure in serration shown by penetrant-oil inspection.